REMARKS:

This communication is in response to the detailed Office Action mailed March 21, 2005. The Examiner rejected claims 1, 3-6 and 9-12. Applicant has amended claims 1 and 9. The Examiner's comments and rejections are addressed below:

35 U.S.C. §112, ¶2 Rejection

The Examiner rejected claims 1 and 3-5, under 35 U.S.C. § 112, second paragraph as being indefinite for failing to particularly point out and distinctly claim the subject matter which Applicant regards as the invention. More specifically, the Examiner stated that the recitation of "a lever diagram" is unclear as a structural element that sequentially provides occupational positions for the operational elements of the planetary gear sets. However, Applicant respectfully traverses the rejection, in light of the amendments.

Applicant has amended independent claim 1 by deleting the limitation of "occupying sequential positions in a lever diagram" on lines 3-10 of the claim. Applicant believes this amendment removes any indefiniteness in claim 1. Because claims 3-5 depend on claim 1, they are no longer indefinite, in light of the above amendment. Therefore, based on the foregoing, Applicant respectfully requests the Examiner to remove the § 112 rejection and allow the claims.

35 U.S.C. § 102(a) Rejection

The Examiner rejected claims 6 and 9-12 under 35 U.S.C. 102(a) as being anticipated by WO 03/054420 A1 ("Gumpoltsberger"). However, Applicant respectfully traverses this rejection because the Examiner relied upon an ineffective reference.

In particular, the Examiner improperly relied on Gumpoltsberger because it was filed with the PCT on December 17, 2002 and published on July 3, 2003. Both of these dates for Gumpoltsberger are after December 3, 2002, the priority date of the present application. The present application claims priority to Korean Application No. 2002-0076294. This right of priority is evidenced by the filing of a certified copy of the original Korean application with the present application on September 24, 2003. In further support, Applicant encloses a copy of the translated Korean application under Exhibit A and certifies that the enclosed document

represents an accurate translation of the certified copy of Korean Application No. 2002-0076294.

Based on the above, the Examiner cannot assert Gumpoltsberger as prior art against the present application, and Applicant respectfully requests withdrawal of this rejection for claims 6 and 9-12.

35 U.S.C. § 102(e) Rejection

The Examiner rejected claims 9, 10, and 12, under 35 U.S.C. § 102(e), as being anticipated by U.S. Patent No. 6,669,595 ("Raghavan"). However, Applicant respectfully traverses this rejection, in light of the amendments.

Applicant has amended claim 9 to recite the first sun gear being fixed to the second sun gear, the first pinion being fixed to the third ring gear, the first ring gear being variably connected with the third pinion carrier, the input shaft being fixed to the first and second sun gears and variably connected with the third pinion carrier, and the output shaft is fixed to the first pinion carrier and the third ring gear. Support for the amendment can be found in various portions of the specification, such as Paragraph [0030] and FIG. 1. Furthermore, this amendment does not add new matter to claim 9.

Raghavan teaches an arrangement of planetary gear sets for a transmission that can be utilized in powertrains to provide at least seven forward speed ratios and one reverse speed ration (Abstract and Figs. 1a, 2a, 3a, 4a, 5a, 6a, 7a, 8a, 9a, 10a, 11a, 12a, and 13a). The planetary gear sets include a sun gear member, a ring gear member, and a planet carrier assembly, in which the planet carrier assembly further includes a plurality of pinion gears (see, e.g., col. 22, lines 42-46).

However, Raghavan does not teach each of the limitations of claim 9, which is best illustrated through Fig. 9a. For example, Raghavan does not teach a first sun gear being fixed to the second sun gear because the first sun gear (822) and second sun gear (842) of Raghavan are fixed to their respective torque transmitting mechanisms (852 and 854) rather than each other. Raghavan also does not provide a first pinion carrier being fixed to the third ring gear because it has an interconnecting member (872) separating the first pinion carrier (829) and the third ring gear (834) from each other, thereby not fixing these items to each other. The first ring gear (824) and third pinion carrier (839) of Raghavan are not variably connected with each other because the interconnecting member (870) always connects these

parts. Raghavan does not teach an input shaft being fixed to the first and second sun gears and variably connected with the third pinion carrier because it only discloses an input shaft (17) that is fixed to the respective torque mechanisms of the first and second sun gears (822 and 842), as well being fixed to a torque transmitting mechanism that is fixed to the planetary gear set (830) that is also fixed to the third pinion carrier. The output shaft of Raghavan is not fixed to the first pinion carrier and the third ring gear because it only provides that the third ring gear is fixed to the interconnecting member (870) that is fixed to the first pinion (829), which is then fixed to the plurality of pinion gears (827) that is also fixed to the output shaft. Moreover, Raghavan does not teach the use of an always stationary second pinion carrier because a plurality of pinion gears (847) are rotatably mounted to the second pinion carrier (849), which means that the second pinion carrier (849) cannot always be stationary.

In light of the above, Raghavan does not teach all of the limitations of claim 9 and, therefore, cannot anticipate claim 9. Because claims 10 and 12 depend on claim 9, Raghavan also does not anticipate them. Based on the foregoing, Applicant respectfully requests withdrawal of this rejection.

Allowable Subject Matter

The Examiner objected to claim 8 as being dependent upon a rejected base claim but would be allowable if rewritten in independent form including all of the limitations of the base claim and any intervening claims. Yet, as Applicant has stated earlier, the Examiner improperly rejected claim 6, from which claim 8 depends on. Based on this situation, claim 8 is allowable as it is, without amending it into independent form.

The Examiner also noted that claims 1 and 3-5 would be allowable if rewritten or amended to overcome the rejection under 35 U.S.C. § 112, second paragraph. As mentioned above, Applicant has amended claim 1, so that claim 1 and its dependents, claims 3-5, now overcome the rejection and should be allowable.

Conclusion

In light of the present amendments and the above arguments, the Applicants believe claims 1, 3-6, and 8-12 are now allowable and the rejections moot. Should the Examiner have any continuing objections or concerns, the Examiner is respectfully asked to contact the undersigned at 415-442-1000 in order to expedite allowance of this case. Authorization is granted to charge any outstanding fees due at this time for the continued prosecution of this matter to Morgan, Lewis & Bockius LLP Deposit Account No. 50-0310 (matter no. 060944-0134).

Respectfully submitted,

Date June 21, 2005

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Exhibit A: Translation of Korean Application No. 2002-0076294

SIX-SPEED POWERTRAIN OF AN AUTOMATIC TRANSMISSION OF A VEHICLE

FIELD OF THE INVENTION

Generally, the present invention relates to an automatic transmission. More particularly, the present invention relates to a powertrain of an automatic transmission that realizes multiple shift speeds with a combination of a plurality of planetary gear sets.

BACKGROUND OF THE INVENTION

A typical shift mechanism of an automatic transmission utilizes a combination of a plurality of planetary gear sets. A powertrain of such an automatic transmission that includes the plurality of planetary gear sets changes rotating speed and torque received from a torque converter of the automatic transmission and accordingly changes and transmits the changed torque to an output shaft.

It is well known that when a transmission realizes a greater number of shift speeds, speed ratios of the transmission can be more optimally designed and therefore a vehicle can have better fuel mileage and better performance. For that reason, an automatic transmission that enables more shift speeds is under constant investigation.

In addition, with the same number of speeds, features of a powertrain such as durability, efficiency in power transmission, and size depend a lot on the layout of combined planetary gear sets. Therefore, designs for a combining

structure of a powertrain are also under constant investigation.

A manual transmission that has too many speeds causes inconvenience of excessively frequent shifting operations to a driver. Therefore, the positive features of more shift-speeds are more important for automatic transmissions because an automatic transmission automatically controls shifting operations basically without needing manual operation.

In addition to various developments regarding four and five speed powertrains, powertrains of automatic transmissions realizing six forward speeds and one reverse speed have recently been introduced, examples of which are found in U.S. patent US6,071,208 that was issued on June 6, 2000, and in U.S. patent US5,226,862 that was issued on July 13, 1993.

FIG. 1 illustrates a powertrain of the patent US6,071,208, and FIG. 2 shows an operational chart for the powertrain.

Referring to FIG. 1, the powertrain of the patent US6,071,208 includes a double pinion planetary gear set PG1 and a pair of single pinion planetary gear sets PG2 and PG3. A first carrier 4 is fixedly connected to an input shaft 2, and a second carrier 22 always operates as an output element.

Regarding connections between operational elements, a first ring gear 6 and a third ring gear 8, a second sun gear 12 and a third sun gear 10, and a second ring gear 16 and a third carrier 14 are fixedly interconnected, respectively.

Meanwhile, the first carrier 4 is variably connected to a first sun gear 18 and the third carrier 14 interposing a first clutch C1 and a second clutch C2, respectively.

In addition, the powertrain further includes a first brake B1 that can stop rotation of the fixedly connected second and third sun gears 12 and 10, a second brake B2 that can stop rotation of the third carrier 14, a third brake B3 that can stop rotation of the first and third ring gears 6 and 8, and a fourth brake B4 that can stop the first sun gear 18.

As described above, the six-speed powertrain of US6,071,208 includes six friction elements of two clutches and four brakes. However, it is preferable to use fewer friction elements to enable six forward speeds and one reverse speed so that an automatic transmission can be more light and compact.

FIG. 2 is an operational chart for the powertrain of US6,071,208, and FIG. 3A-3F are charts showing operation states obtained when the powertrain is operated according to the operational chart in FIG. 2.

In particular, FIG. 3A shows detailed specifications of the powertrain of US6,071,208, i.e., gear ratios of each planetary gear set. FIG. 3B shows speed ratios in each shift-speed of the powertrain obtained by the detailed specification of FIG. 3A. In addition, FIG. 3C shows rotation speeds of each operational element relative to that of the input element, for each shift-speed. FIG. 3D shows slip speeds of friction elements at each shift-speed. FIG. 3E shows torque loads that each operational element or each friction element undertakes. FIG. 3F shows planetary gear sets that take part in power transmission in each shift-speed.

As shown in FIG. 2, the powertrain of US6,071,208 operates the first and fourth brakes B1 and B4 at a first speed, the first clutch C1 and the first brake B1 at a second speed, the second clutch C2 and the first brake B1 at a

third speed, the first and second clutches C1 and C2 at a fourth speed, the second clutch C2 and the fourth brake B4 at a fifth speed, and the second clutch C2 and the third brake B3 at a sixth speed, respectively. The second and fourth brakes B2 and B4 are operated at a reverse speed.

Referring to the operational chart, the operation state of each operational element of the powertrain of US6,071,208 is described in detail. The planetary gear sets of the powertrain are supposed to have gear ratios shown in FIG. 3A such that the speed ratios shown in FIG. 3B are achieved.

(1) At the third forward speed, the first sun gear 18 rotates at a speed of more than twice that of the rotation speed of the input shaft (refer to FIG. 3C). In addition, the slip speed of the fourth brake B4, which is not operated in the third speed, becomes as high as that of the first sun gear 18 (refer to FIG. 3D).

The third forward speed is frequently engaged in the case that acceleration is needed, since a six-speed automatic transmission usually achieves the speed ratio of 1:1 at the fourth forward speed. Therefore, durability of an automatic transmission deteriorates if an element always rotates at a high speed in such a shift speed.

(2) Referring to FIG. 3D, slip speeds of friction elements are excessive for all speed ranges, which deteriorates durability of an automatic transmission and also causes excessive power loss. Therefore, the powertrain should be improved to have lesser slip speeds of friction elements for speeds D2-D6.

In particular, the sum of slip speeds of friction elements becomes excessively large at the sixth forward speed D6, and therefore, the durability problem is at its maximum at the sixth forward speed.

(3) Referring to FIG. 3F, when considering the number of planetary gear sets that take part in power transmission, at least two planetary gear sets take part in the power transmission for the fifth and sixth speeds, which deteriorates power efficiency. It is preferable that efficiency of power transmission is improved.

FIG. 4 illustrates a powertrain of the patent US5,226,862, and FIG. 5 shows an operational chart for the powertrain.

Referring to FIG. 4, the six speed powertrain of US5,226,862 includes a double pinion planetary gear set G1 and a pair of single pinion planetary gear sets G2 and G3. The input shaft I is variably connected to second and third sun gears S2 and S3 that are fixedly interconnected, a second carrier PC2, and a first sun gear S1, through first, second, and third clutches C1, C2, and C3, respectively. A third carrier PC3 always operates as an output element, and a first carrier PC1 is fixed to a transmission housing 7.

Regarding connections between operational elements, first and second ring gears R1 and R2, the second carrier PC2 and a third ring gear R3, and the second and the third sun gears S2 and S3 are fixedly interconnected, respectively.

The powertrain further includes a first brake B1 that can stop rotation of the fixedly interconnected second carrier PC2 and third ring gear R3, and a second brake that can stop the fixedly interconnected first and second ring gears R1 and R2.

FIG. 5 is an operational chart for the powertrain of US5,226,862, and FIGs. 6A-6F are charts showing operation states obtained when the powertrain

is operated according to the operational chart in FIG. 5.

In particular, FIG. 6A shows detailed specifications of the powertrain of US5,226,862, i.e., gear ratios of each planetary gear set. FIG. 6B shows speed ratios in each shift-speed of the powertrain obtained by the detailed specification of FIG. 6A. FIG. 6C shows rotation speeds of each operational element relative to that of the input element, for each shift-speed. FIG. 6D shows slip speeds of friction elements at each shift-speed. FIG. 6E shows torque loads that each operational element or each friction element undertakes. FIG. 6F shows planetary gear sets that take part in power transmission in each shift-speed.

As shown in FIG. 5, the powertrain of US5,226,862 operates the first clutch C1 and the first brake B1 at a first speed, the first clutch C1 and the second brake B2 at a second speed, the first clutch C1 and the third clutch C3 at a third speed, the first and the second clutches C1 and C2 at a fourth speed, the second and third clutches C2 and C3 at a fifth speed, and the second clutch C2 and the second brake B2 at a sixth speed, respectively. The third clutch C3 and the first brake B1 are operated at a reverse speed.

Referring to the operational chart, the operation state of each operational element of the powertrain of US5,225,862 is shown in FIGs. 6A-6F. The planetary gear sets of the powertrain are supposed to have gear ratios shown in FIG. 6A such that the speed ratios shown in FIG. 6B are achieved. Shaded numbers indicate disadvantages of the powertrain.

In particular, at the fourth speed, the first sun gear S1 rotates at a speed of more than twice that of the rotation speed of the input shaft (refer to FIG. 6C).

The third forward speed is frequently engaged in the case that acceleration is needed, since a six-speed automatic transmission usually achieves the speed ratio of 1:1 at the fourth forward speed. Therefore, durability of an automatic transmission deteriorates if an element always rotates at a high speed in such a shift-speed.

Furthermore, at the fourth speed, the speed of a first planetary gear (i.e., pinion gear of the first planetary gear set) P1 is almost 3.8 times that of the input shaft (refer to FIG. 6C).

This kind of high relative revolution speed may cause critical harm to durability of planetary gear sets. So, it is believed that the powertrain of US5,226,862 is not appropriate for use for automotive vehicles unless the durability is substantially supplemented.

The information disclosed in this Background of the Invention section is only for enhancement of understanding of the background of the invention, and should not be taken as an acknowledgement or any form of suggestion that this information forms the prior art that is already known to a person skilled in the art.

SUMMARY OF THE INVENTION

An exemplary powertrain of an automatic transmission useful with the present invention includes first, second, and third planetary gear sets.

The first planetary gear set has first, second, and third operational elements, and the first, second, and third operational elements occupy sequential positions in a lever diagram.

The second planetary gear set has fourth, fifth, and sixth operational elements, and the fourth, fifth, and sixth operational elements occupy sequential positions in a lever diagram.

The third planetary gear set has seventh, eighth, and ninth operational elements, and the seventh, eighth, and ninth operational elements occupy sequential positions in a lever diagram.

The first operational element is fixedly connected to the fourth operational element and always receives an input torque. The second operational element is fixedly connected to the seventh operational element and always outputs an output torque. The third operational element is variably connected to either of the eighth operational element or the ninth operational element via a second clutch. The fifth operational element is variably connected to the ninth operational element via a first clutch. The sixth operational element is always stationary. The eighth operational element is variably connected to an input shaft via a third clutch and is subject to a stopping operation of a first brake. The ninth operational element is subject to a stopping operation of a second brake.

In a further embodiment, the third operational element is variably connected to the eighth operational element via the second clutch.

It is preferable that the first and second planetary gear sets are single pinion planetary gear sets; the first, second, and third operational elements are respectively a sun gear, a carrier, and a ring gear of the first planetary gear set; and the fourth, fifth, and sixth operational elements are respectively a ring gear, a carrier, and a sun gear of the second planetary gear set.

It is also preferable that the third planetary gear set is a double pinion planetary gear set, and the seventh, eighth, and ninth operational elements are respectively a sun gear, a ring gear, and a carrier of the third planetary gear set.

It is also preferable that the first, second, and third planetary gear sets are arranged in the order of the first, third, and second planetary gear sets.

BRIEF DESCRIPTION OF THE DRAWINGS

The accompanying drawings, which are incorporated in and constitute a part of the specification, illustrate an embodiment of the invention, and, together with the description, serve to explain the principles of the invention:

FIG. 1 illustrates an exemplary six-speed powertrain according to the prior art;

FIG. 2 is an operational chart for the powertrain shown in FIG. 1;

FIGs. 3A-3F are charts showing operation states obtained when the powertrain shown in FIG. 1 is operated according to the operational chart in FIG. 2;

FIG. 4 illustrates another exemplary six-speed powertrain according to the prior art;

FIG. 5 is an operational chart for the powertrain shown in FIG. 4;

FIGs. 6A-6F are charts showing operation states obtained when the powertrain shown in FIG. 4 is operated according to the operational chart in FIG. 5;

FIG. 7 illustrates a powertrain of an automatic transmission according to a preferred embodiment of the present invention;

FIG. 8 is an operational chart for a powertrain of an automatic transmission according to a preferred embodiment of the present invention;

FIG. 9 is a lever diagram showing nodes N1-N5 of a powertrain of an automatic transmission according to a preferred embodiment of the present invention;

FIG. 10 is a speed diagram of a powertrain of an automatic transmission according to a preferred embodiment of the present invention; and

FIGs. 11A-11F are charts showing operation states of a power train of an automatic transmission according to a preferred embodiment of the present invention.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

A preferred embodiment of the present invention will hereinafter be described in detail with reference to the accompanying drawings.

FIG. 7 illustrates a powertrain of an automatic transmission according to a preferred embodiment of the present invention.

As shown in FIG. 7, a power train of an automatic transmission of the present invention includes first, second, and third planetary gear sets PG1, PG2, and PG3.

The first planetary gear set is a single pinion simple planetary gear set, and includes a first sun gear S1, a first pinion carrier (called "carrier" hereinafter) PC1, and a first ring gear R1.

The second planetary gear set is a double pinion simple planetary gear set, and includes a second sun gear S2, a second carrier PC2, and a first ring

gear R2.

The third planetary gear set is a single pinion simple planetary gear set, and includes a third sun gear S3, a third carrier PC3, and a third ring gear R3.

The first, second, and third planetary gear sets PG1, PG2, and PG3 are arranged in the order of the first, third, and second planetary gear sets PG1, PG3, and PG2, from an input shaft.

The first sun gear S1 and the second sun gear S2 are fixedly connected to the input shaft, respectively. The third carrier PC3 is connected to the input shaft interposing a third clutch C3.

The first carrier PC1 and the third ring gear R3 are fixedly interconnected, and the first carrier PC1 always operates as an output element.

The first ring gear R1 and the third carrier PC3 are variably interconnected via a second clutch C2. The third sun gear S3 and the second ring gear R2 are variably interconnected via a first clutch C1.

The powertrain of the present embodiment further includes a first brake B1 that can stop the rotation of the third carrier PC3, and a second brake B2 that can stop the rotation of the third sun gear S3. The second carrier PC2 of the second planetary gear set PG2 is fixed to the transmission housing such that it is always stationary.

FIG. 8 is an operational chart for a powertrain of an automatic transmission according to a preferred embodiment of the present invention.

As shown in FIG. 8, the powertrain of the present embodiment enables forward six speeds plus reverse one speed by operating the second clutch C2 and the first brake B1 at a first forward speed D1, the second clutch C2 and the

second brake B2 at a second speed D2, the first clutch C1 and the second clutch C2 at a third speed D3, the second clutch C2 and the third clutch C3 at a fourth speed D4, the first clutch C1 and the third clutch C3 at the fifth speed D5, the third clutch C3 and the second brake B2 at a sixth speed D6, and the first clutch C1 and the first brake B1 at a reverse speed R.

Shifting operation of the powertrain of the present embodiment according to the operational chart shown in FIG. 8 is hereinafter described in detail with reference to its lever diagram and speed diagram.

FIG. 9 is a lever diagram showing nodes N1-N5 of a powertrain of an automatic transmission according to a preferred embodiment of the present invention.

As shown in FIG. 9, the first sun gear S1, the second carrier PC1, and the first ring gear R1 of the first planetary gear set PG1 are sequentially located in the lever diagram, and are denoted as operating nodes N1, N2, and N3

The third ring gear R3, the third carrier PC3, and the third sun gear S3 of the third planetary gear set PG3 are also sequentially located in the lever diagram. The third ring gear R3 and the third carrier PC3 respectively correspond to the second and third operating nodes N2 and N3, and the third sun gear S3 is denoted as a new operating node N4 shown to the right of the node N3.

The second sun gear S2, the second ring gear R2, and the second carrier PC2 of the second planetary gear set PG2 are also sequentially located in the lever diagram. The second sun and ring gears S2 and R2 respectively correspond to the first and fourth nodes N1 and N4, and the second carrier PC2

is denoted as a new operating node N5 shown to the right of the node N4.

Each operating node is positioned based on the specification of the planetary gear sets shown in FIG. 11A. For better comprehension, the lever for the second planetary gear set PG2 is shown in a scale different from that of the first and third planetary gear sets PG1 and PG3.

FIG. 10 is a speed diagram of a powertrain of an automatic transmission according to a preferred embodiment of the present invention. In more detail, the speed diagram is drawn based on the gear ratios of the planetary gear sets as shown in FIG. 11A.

As described above, the second sun gear S2 of the second planetary gear set PG2 is fixedly connected to the input shaft, and the second carrier PC2 is always stationary. Therefore, the speed line for the second planetary gear set PG2 also becomes stationary (refer to a dotted line in FIG. 10).

In addition, the first sun gear S1 of the first planetary gear set PG1 is fixedly connected to the input shaft. Therefore, the first operating node N1 (i.e., operational elements correspondent thereto) rotates at the same speed as the input shaft.

In addition, the first planetary gear set PG1 and the third planetary gear set PG3 form four nodes, i.e., first, second, third, and fourth operating nodes N1, N2, N3, and N4 during the forward first through fourth speeds D1-D4, because the second clutch C2 remains in operation at those speeds (refer to FIG. 9).

Having the second clutch C2 operated, shifting operation during first through fourth speeds D1-D4 of the powertrain of the present embodiment is as

follows.

At the first speed, the rotation speed of the third node N3 becomes zero (0) because the first brake B1 is operated. Therefore, the rotation speed at the first speed D1 of the first carrier PC1 (the output element) is found by a first speed line designated by “D1”.

At such a first speed, the first sun gear S1 operates as the input element, the first ring gear R1 connected to the third carrier PC3 functions as a reaction element, and the first carrier PC1 functions as an output element. Therefore, only the first planetary gear set PG1 takes part in the power transmission at the first speed.

At the second speed, the first brake B1 is released and the second brake B2 operates. Now, the third sun gear S3 (an element at the fourth node N4) stops its rotation by the operation of the second brake B2. Therefore, the rotation speed of the output element PC1 at the second speed is found by a second speed line designated by “D2”, and the second speed line shows that the output element PC1 rotates faster at the second speed than at the first speed.

At such a second speed, the first and third planetary gear sets PG1 and PG3 take part in the power transmission.

At the third speed, the first clutch C1 operates in addition to the operation of the second clutch C2. Therefore, the third sun gear S3 and the second ring gear R2 on the fourth node N4 rotate at the same speed. Accordingly, at this third speed, all the operating elements in the first, second, and third planetary gear sets PG1-PG3 lie on the same speed line designated

by “D3” in FIG. 10.

At such a third speed, all the first, second, and third planetary gear sets PG1, PG2, and PG3 take part in the power transmission.

At the fourth speed, the third clutch C3 operates. In this case, the third carrier PC3 rotates with the input shaft, and the first ring gear R1 that is connected to the third carrier PC3 by the currently operating second clutch C2 also rotates with the input shaft. As a result, the first and third planetary gear sets PG1 and PG3 rotate as a whole (i.e., without a relative movement of elements in the gear sets PG1 and PG3) with the input shaft.

Therefore, a fourth speed line for the first and third planetary gear sets PG1 and PG3 at the fourth speed becomes a horizontal line designated by “D4” in FIG. 10.

At such a fourth speed, the gear sets PG1 and PG3 rotate as a whole, and the power is directly transmitted from the input shaft to the output shaft without causing a relative movement or rotation of elements in the gear sets PG1 and PG3. Therefore, no planetary gear set effectively takes part in the power transmission.

Now the shifting operation at the fifth, sixth, and reverse speeds is described in detail.

At the fifth, sixth, and reverse speeds, the second clutch C2 is released such that speed lines for the first planetary gear set PG1 and the third planetary gear set PG3 are separated.

At the fifth speed, the first clutch C1 is operated, and accordingly, the third sun gear S3 of the third planetary gear set PG3 rotates with the second

ring gear R2 of the second planetary gear set PG2. In addition, the third clutch C3 is operated, and accordingly, the third carrier PC3 of the third planetary gear set PG3 rotates with the input shaft.

Therefore, the third ring gear R3 rotates faster than the input shaft. The first carrier PC1 (i.e., the output element) fixedly connected to the third ring gear R3 also rotates with the third ring gear R3, i.e., faster than the input shaft (refer to speed lines designated by "D5" in FIG. 10).

At such a fifth speed, the first planetary gear set PG1 does not take a torque load for the power transmission: the second and third planetary gear sets PG2 and PG3 take the load for the power transmission.

At the sixth speed, the third carrier PC3 of the third planetary gear set PG3 rotates with the input shaft due to the operation of the third clutch C3, as it does at the fifth speed.

However, the third sun gear S3 of the third planetary gear set PG3 stops its rotation because the second brake B2 is operated. In this case, the speed line for the third planetary gear set PG3 at the sixth speed is rotated clockwise from that at the fifth speed.

Therefore, the third ring gear R3 rotates faster at the sixth speed than at the fifth speed, and accordingly, the first carrier PC1 directly connected to the third ring gear R3 also rotates faster than at the fifth speed (refer to speed lines designated by "D6" in FIG. 10).

At such a sixth speed, the first planetary gear set PG1 does not take a torque load for the power transmission, only the third planetary gear set PG3 takes the load for the power transmission.

At the reverse speed, the first clutch C1 is operated, and accordingly, the third sun gear S3 of the third planetary gear set PG3 rotates with the second ring gear R2 of the second planetary gear set PG2. In addition, the first brake B1 is operated such that the third carrier PC3 stops its rotation, and therefore, the third ring gear R3 has a negative rotation speed, i.e., rotation opposite to that of the input shaft.

Therefore, the first carrier PC1 fixedly connected to the third ring gear R3 also has negative rotation speed, and accordingly the reverse speed is achieved.

At such a reverse speed, the first planetary gear set PG1 does not take torque load for the power transmission: the second and third planetary gear sets PG2 and PG3 take the load for the power transmission.

FIGs. 11A-11F are charts showing operation states of a power train of an automatic transmission according to a preferred embodiment of the present invention.

In particular, FIG. 11A shows detailed specifications of the powertrain of the present embodiment, i.e., gear ratios of each planetary gear set. FIG. 11B shows speed ratios in each shift-speed of the powertrain of the present embodiment obtained by the detailed specification of FIG. 11A. FIG. 11C shows rotation speeds of each operational element relative to that of the input element, for each shift-speed. FIG. 11D shows slip speeds of friction elements at each shift-speed. FIG. 11E shows torque loads that each operational element or each friction element undertakes. FIG. 11F shows planetary gear sets that take part in power transmission in each shift-speed.

Details shown in FIG. 11F are obvious from the above description of shifting operation of the powertrain of the present invention, and the numbers shown in FIGs. 11C-11E may be obviously calculated based on the structural features and operational chart of the powertrain of the present embodiment.

According to the powertrain of the present embodiment, no operational element rotates faster than the input shaft at the third speed that is frequently engaged for acceleration (refer to FIG. 11C), and therefore, slip speeds of friction elements not operated at the third speed are less than the rotation speed of the input shaft (refer to FIG. 11D).

When the numerals shown in FIG. 11D are compared with those in FIGs. 3D and 6D, it is apparent that the powertrain of the present embodiment shows less slip speeds of friction elements overall at the second to sixth speeds than the powertrains of US6,071,208 and US5,226,862.

It is well known that more planetary gear sets implies more loss of power during power transmission. When the numerals shown in FIG. 11F are compared with those in FIGs. 3F and 6F, it is apparent that the powertrain of the present embodiment has less planetary gear sets involved in the power transmission at many of the shift-speeds and accordingly shows better power efficiency.

According to a preferred embodiment of the present invention, six forward speeds and one reverse speed are achieved with a minimized number of friction elements such that an automatic transmission becomes light and compact. Durability is increased due to reduction of rotation speeds of operational elements at a shift-speed frequently engaged for acceleration. A

further increase of durability and reduction of power loss is also achieved by reduction of slip speeds of friction elements. A shortened route of power transmission also contributes to an increase of durability and reduction of power loss.

In addition, dominant usage of single pinion planetary gear sets also contributes to reduction of power loss.

In addition, layout of the output element toward the input shaft enables easy installation of an automatic transmission for a front-wheel drive vehicle.

Furthermore, the torque steer effect of a front-wheel drive vehicle can be reduced because the length difference of left and right drive shafts is reduced since the planetary gear set directly connected to the output shaft is positioned toward the input shaft.

While this invention has been described in connection with what is presently considered to be the most practical and preferred embodiment, it is to be understood that the invention is not limited to the disclosed embodiments, but, on the contrary, is intended to cover various modifications and equivalent arrangements included within the spirit and scope of the appended claims.

WHAT IS CLAIMED IS:

1. A powertrain of an automatic transmission, comprising:

a first planetary gear set having first, second, and third operational elements, the first, second, and third operational elements occupying sequential positions in a lever diagram;

a second planetary gear set having fourth, fifth, and sixth operational elements, the fourth, fifth, and sixth operational elements occupying sequential positions in a lever diagram; and

a third planetary gear set having seventh, eighth, and ninth operational elements, the seventh, eighth, and ninth operational elements occupying sequential positions in a lever diagram,

wherein:

the first operational element is fixedly connected to the fourth operational element and always receives an input torque;

the second operational element is fixedly connected to the seventh operational element and always outputs an output torque;

the third operational element is variably connected to either of the eighth operational element and the ninth operational element via a second clutch;

the fifth operational element is variably connected to the ninth operational element via a first clutch;

the sixth operational element is always stationary;

the eighth operational element is variably connected to an input shaft via a third clutch and is subject to a stopping operation of a first brake; and

the ninth operational element is subject to a stopping operation of a

second brake.

2. The powertrain of claim 1, wherein the third operational element is variably connected to the eighth operational element via the second clutch.

3. The powertrain of claim 2, wherein:

the first and second planetary gear sets are single pinion planetary gear sets;

the first, second, and third operational elements are respectively a sun gear, a carrier, and a ring gear of the first planetary gear set; and

the fourth, fifth, and sixth operational elements are respectively a ring gear, a carrier, and a sun gear of the second planetary gear set.

4. The powertrain of claim 2, wherein:

the third planetary gear set is a double pinion planetary gear set; and

the seventh, eighth, and ninth operational elements are respectively a sun gear, a ring gear, and a carrier of the third planetary gear set.

5. The powertrain of claim 2, wherein the first, second, and third planetary gear sets are arranged in the order of the first, third, and second planetary gear sets.

ABSTRACT OF THE DISCLOSURE

The present invention provides a powertrain of an automatic transmission that includes first, second, and third planetary gear sets. The first planetary gear set has first, second, and third operational elements that occupy sequential positions in a lever diagram. The second planetary gear set has fourth, fifth, and sixth operational elements that occupy sequential positions in the lever diagram. The third planetary gear set has seventh, eighth, and ninth operational elements that occupy sequential positions in the lever diagram. The first operational element is fixedly connected to the fourth operational element and always receives an input torque. The second operational element is fixedly connected to the seventh operational element and always outputs an output torque. The third operational element is variably connected to either of the eighth operational element and the ninth operational element via a second clutch. The fifth operational element is variably connected to the ninth operational element via a first clutch. The sixth operational element is always stationary. The eighth operational element is variably connected to an input shaft via a third clutch and is subject to a stopping operation of a first brake. The ninth operational element is subject to a stopping operation of a second brake.